

Chapter – 8 Low Temperature Refrigeration Systems

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It is the general guide line that for industrial refrigeration application, when the system compression ratio exceeds 10 to 15, a compound system might start to look attractive, particularly from the view points of power consumption, annual operating expenses and the pay back; A cascade system might be considered if the evaporative temperature is below -60°F or -80°F . The temperature application range might be overlap widely between various systems. But, at the end, the most effective tool for the comparison is still the system economics.

If the project is to handle a large refrigeration load and the process requires several side loads at different temperature levels, a multistage centrifugal compressor might be the candidate for the project. A multistage compressor can handle the ET down to about -40°F ; A compound multistage centrifugal system can handle the ET down to about -100°F and a cascade multistage centrifugal compressor system can handle the ET down to -250°F .

Compound System:

A compound system is to use two or more compressors, reciprocating, screw or centrifugal, in series using single refrigerant.

A system with multiple compressors in series without any intercooling and desuperheating is considered same as a single compression and it is not considered as a compound system. Therefore, a compound system must include the necessary required components, such as intercooler, economizer or subcooler to improve the system efficiency as described in Chapter 3. The purpose of using a compound system is as the following:

- Increase the refrigeration effect.
- Reduce discharge temperature.
- Reduce the equipment size.
- Reduce the system power consumption.
- Reduce the annual operation expenses.

Most compound systems are 2-stage compression system. A 2-stage system is actually a combination of two single systems with an intermediate flash intercooler or liquid subcooling type intercooler.

The idea intermediate pressure (P_i) of a 2-stage compression is derived from the formula as the following:

$$P_i = \sqrt{P_1 \times P_2}$$

The saturated temperature at the intermediate pressure (P_i) is actually the intermediate temperature of the 2-stage compound system.

Figure 8-1 illustrates how a 2-stage compound with flash intercooling refrigeration system is formed:

Diagram-(1) of the Figure 8-1 is the P-H diagram showing CT, ET and the P_i as indicated in the diagram. The diagram shows that the refrigeration effect δH is small if the system were on a simple single compression cycle.

Diagram-(2) of the Figure 8-1 is the P-H diagram for the low stage circuit of the supposed to be compound refrigeration system. The refrigerant liquid to evaporator is now from the intermediate temperature; the intermediate temperature is considered as the condensing temperature for the low stage compressor. As indicated in the diagram, it is evident that by using flash intercooler for the compound system, the refrigeration effect $\delta H'$ is now increased and became much larger as compared to the δH shown in Diagram-(1) .

Diagram-(3) of the Figure 8-1 is the P-H diagram for the high stage circuit of the supposed to be compound refrigeration system. The intermediate temperature is considered the evaporative temperature for the high stage compressor.

Putting the Diagram-(2) and Diagram-(3) together, it shall be the P-H diagram for this 2-stage screw compound system with flash intercooler as shown in the Diagram-(4) of the Figure 8-1.

The Refrigerant Flow Diagram for this 2-Stage compound system with the flash intercooler is shown in Figure 8-2. The flash intercooler is also providing the desuperheating function for the system.

If the 2-stage compound system is with a shell-and-tube type liquid subcooling intermediate intercooler instead of flash type, the system is shown in Figure 8-3. This type of intercooler provides high pressure subcooled liquid to the evaporator.

The P_i is usually obtained by the idea intermediate pressure formula of square root of ($P_1 \times P_2$). However, it shall be always a good practice to recheck the actual compressor performances to see if the idea P_i yields the optimum efficiencies of both high and low side compressors. Sometimes, the system efficiency can be improved a lot by readjusting the intermediate pressure (P_i) for the actual compressor.

Besides of the intermediate intercooler between the high and low stage, additional intercooler and/or economizer can be used where ever is feasible for the high side and/or the low side compressor to improve the system efficiency.

There is no restriction on what type of compressor to be used for the compound system. However, it is a general practice to use low or medium flow with variable

head compressor (reciprocating or screw) as the high stage; use medium or high flow with constant head compressor (screw or centrifugal) as the low stage.

Multistage Centrifugal System:

Refrigeration system utilizes one multistage centrifugal compressor, no matter how

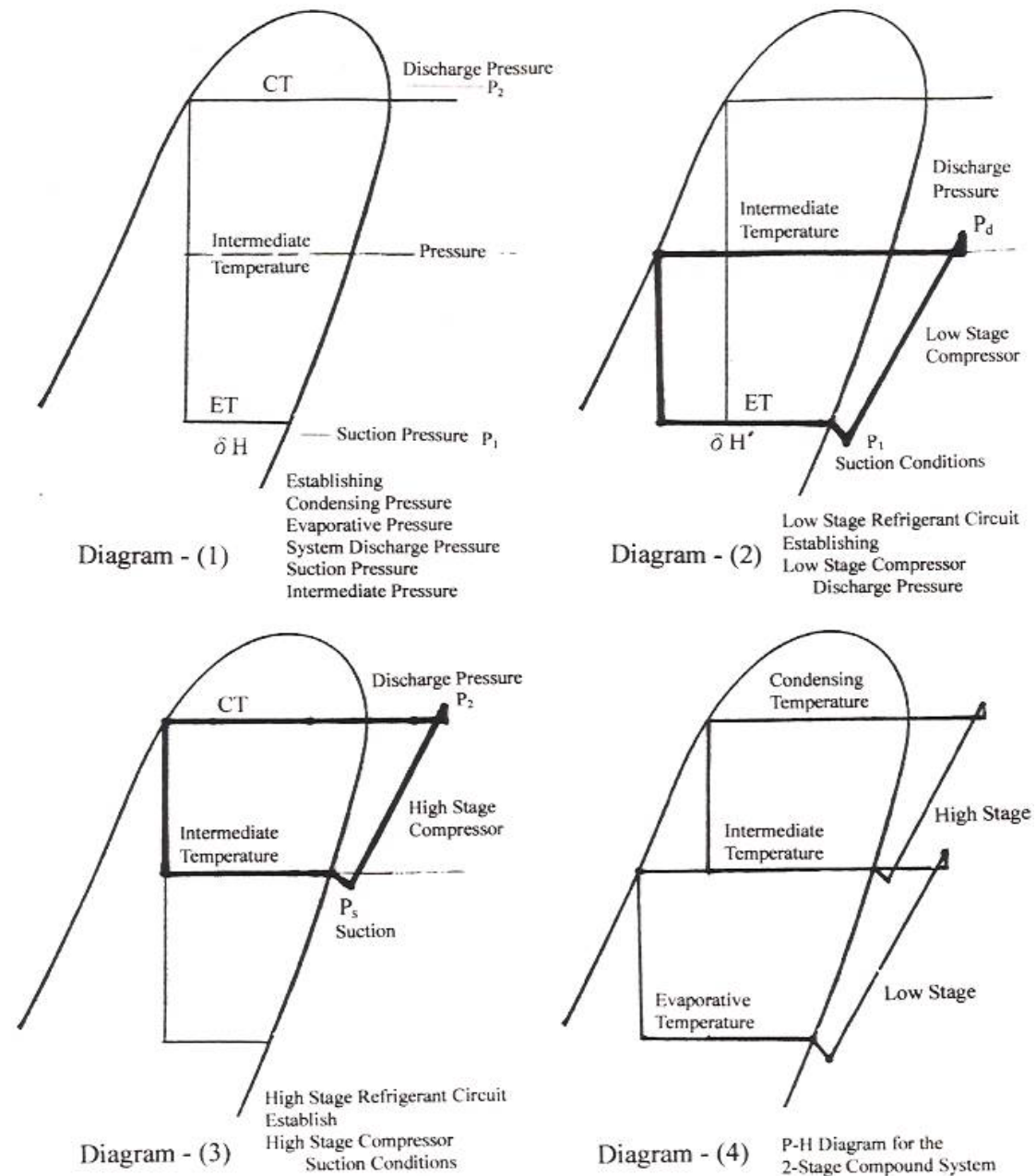


Figure 8-1 Structuring a 2-stage Compound System

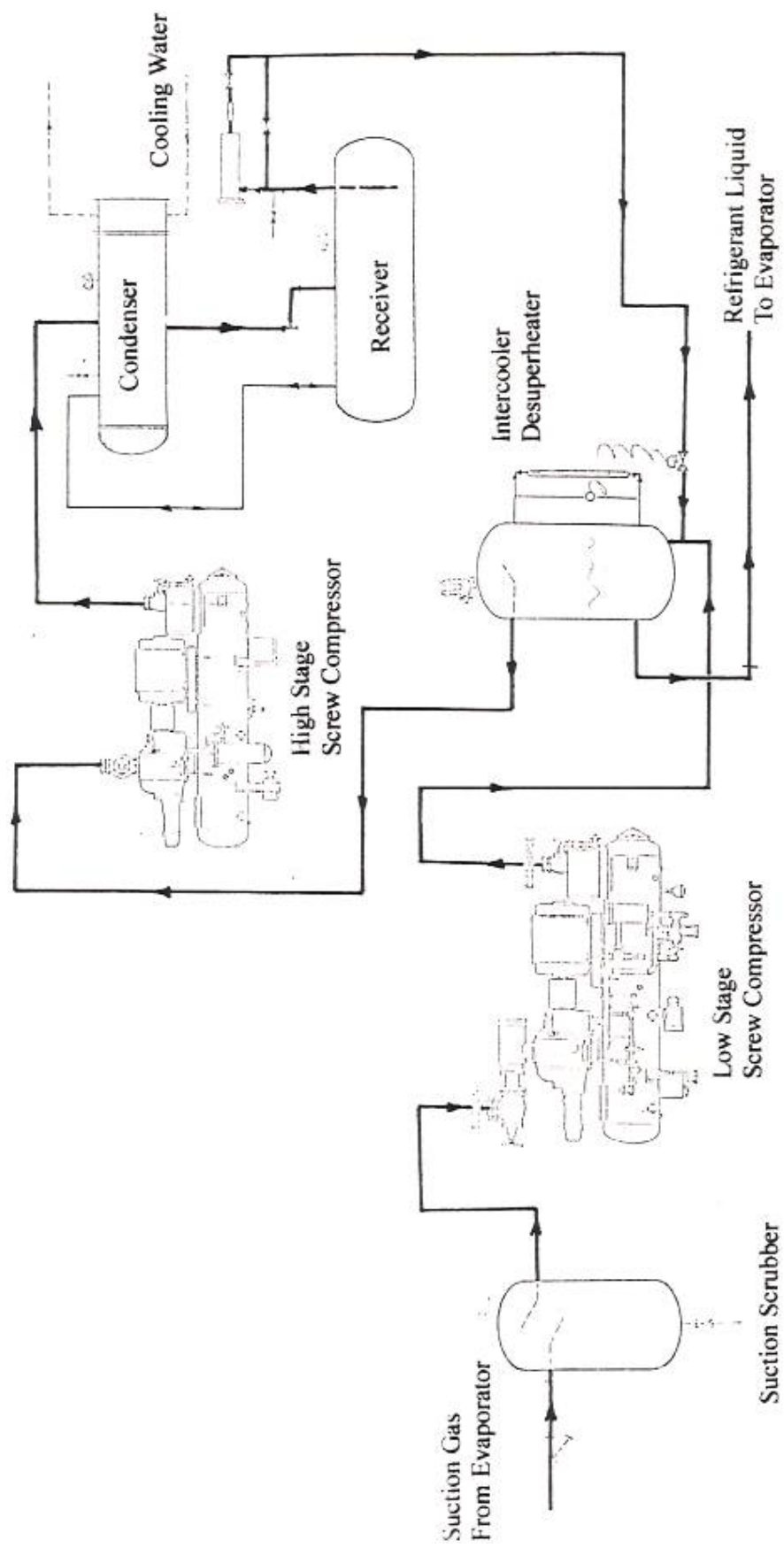


Figure 8-2 Refrigerant Flow Diagram
2-stage Screw Compound System
With Flash Intercooler

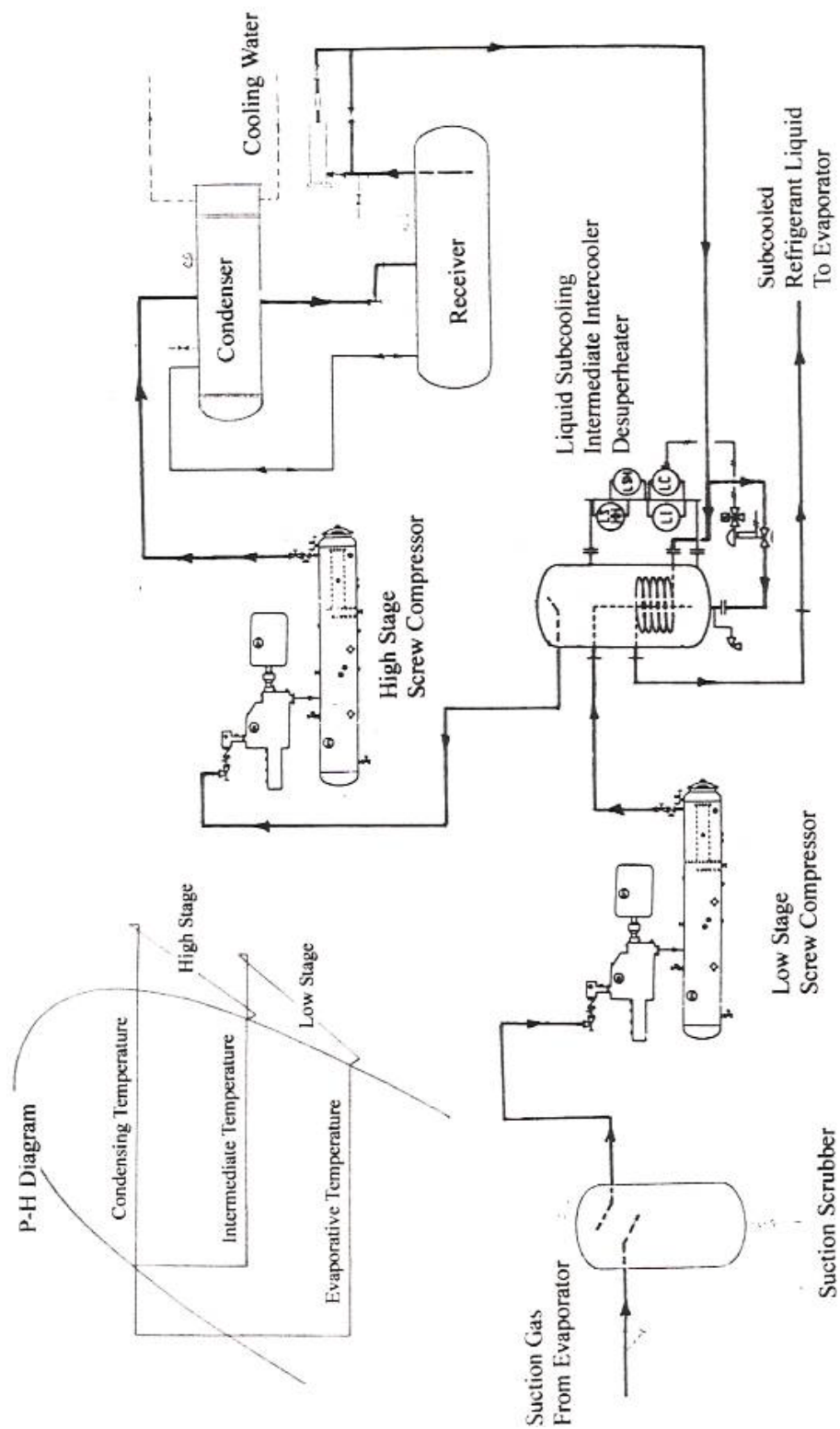


Figure 8-3 Refrigerant Flow Diagram
2-Stage Screw Compound System
With Liquid Subcooling Intercooler

many stages are involved and how complicated the system is, the system is classified as multistage centrifugal system, not compound system, unless the system is with two or more multistage centrifugal compressors.

Figure 8-4 is the refrigerant flow diagram for a refrigeration system with 6-stage centrifugal compressor. The system uses two shell-and-tube type liquid subcoolers to increase the refrigerant effect. The suction gas from the 52°F subcooler is connected to the suction of the 2nd stage impeller of the compressor and the suction gas from the 3°F subcooler is connected to the 4th stage impeller of the compressor. The suction gas from the 35.1°F evaporator is returned to the 3rd stage impeller of the compressor as indicated. The refrigerant use in this case is Propylene (R-1270) which is a commonly used refrigerant for refrigeration systems for hydrocarbon processing industries.

Figure 8-5 is the P-H diagram analysis for this 6-stage centrifugal refrigeration system as shown in Figure 8-4.

Note: The minimum suction gas density for centrifugal compressor when it is used in a compound system is not to be less than 0.01 Lbs/Ft³. The compressor selection might not be feasible if the suction gas density is below this level.

Booster Compound System:

When the low stage compressor of a compound system is designed to handle a smaller head as compared to the high stage compressor; this low stage compressor is called the booster compressor. The reasons that might prompt the use of booster compound are as the following:

- (I) To improve the high stage compressor efficiency.
- (II) To handle evaporative loads at different temperature levels; the intermediate pressure can be set equal to the side load ET and logically it can be handled by the suction of the high stage compressor.
- (III) System economic justifications and other considerations.

Figure 8-6 is a typical booster compound refrigeration system using single stage centrifugal compressor as the booster and a screw compressor as the high stage. The advantage of using this screw and centrifugal combination is that constant head machine (centrifugal) for the booster and variable head machine (screw) for the high stage.

The Figure 8-7 is the P-H diagram analysis for this refrigeration system as shown in Figure 8-6.

Cascade System:

As a general guide line, cascade system might be considered for one or more following reasons:

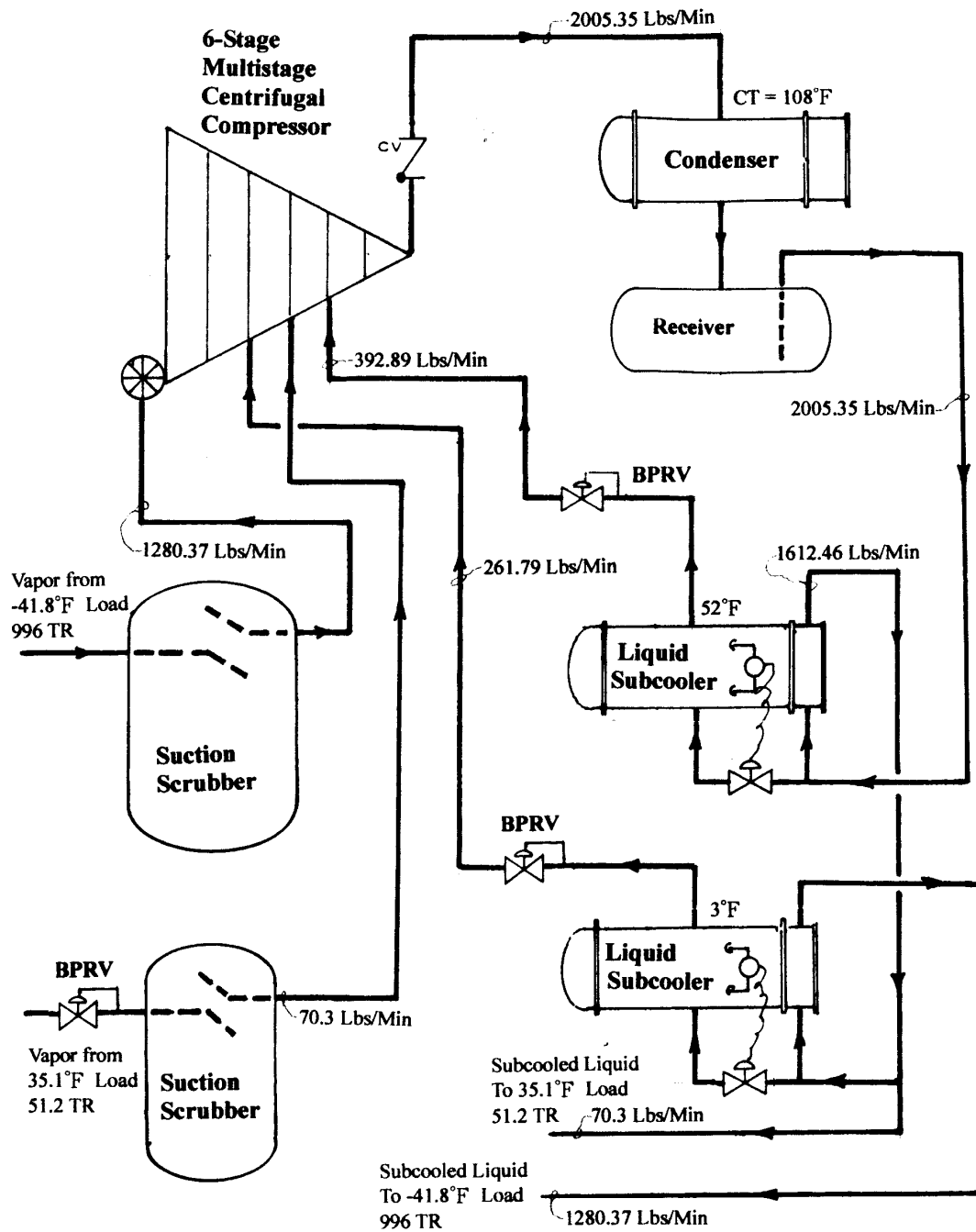


Figure 8-4 R-1270 (Propylene) Refrigerant Flow Diagram
6-Stage Multistage Centrifugal Refrigeration System

R-1270 (PROPYLENE) P-H DIAGRAM ANALYSIS

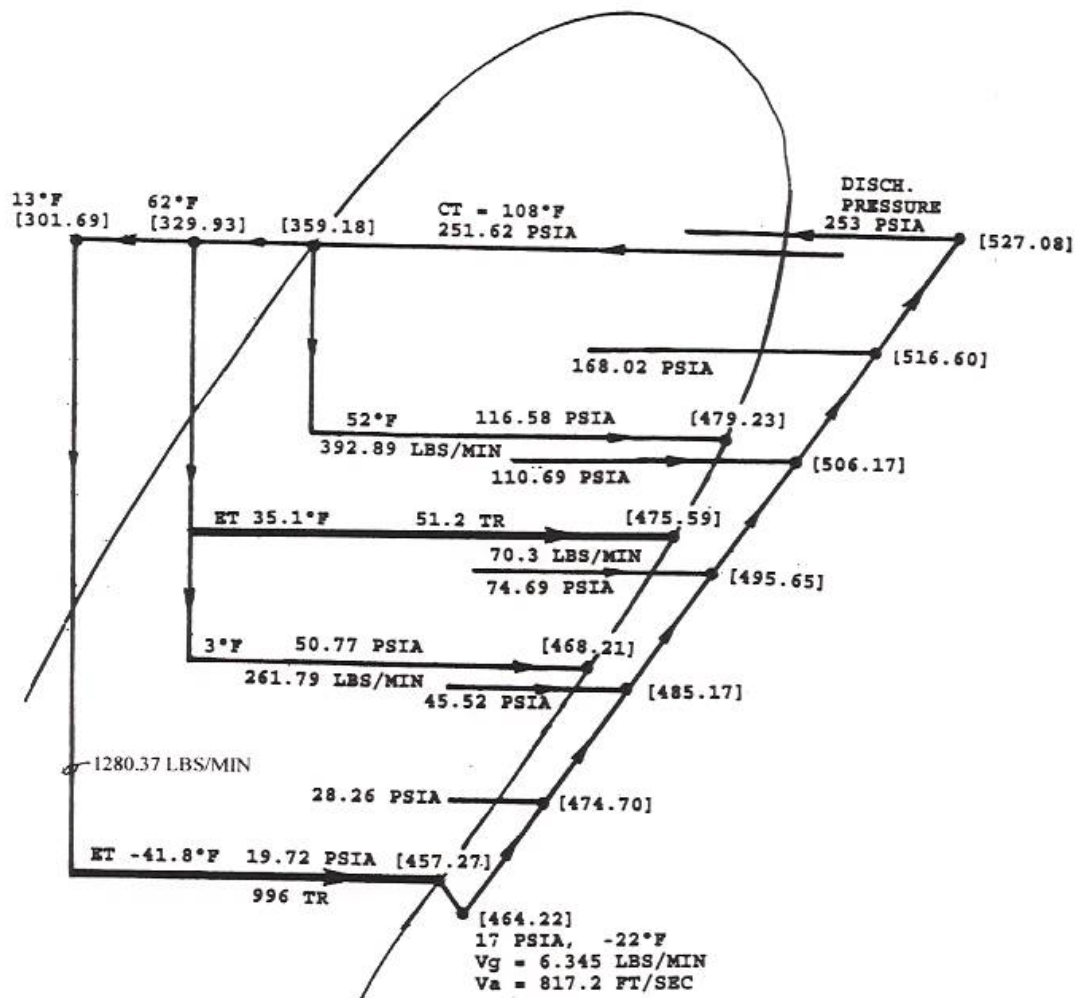


Figure 8-5 R-1270 P-H Diagram
For 6-Stage Multistage Centrifugal
Refrigeration System

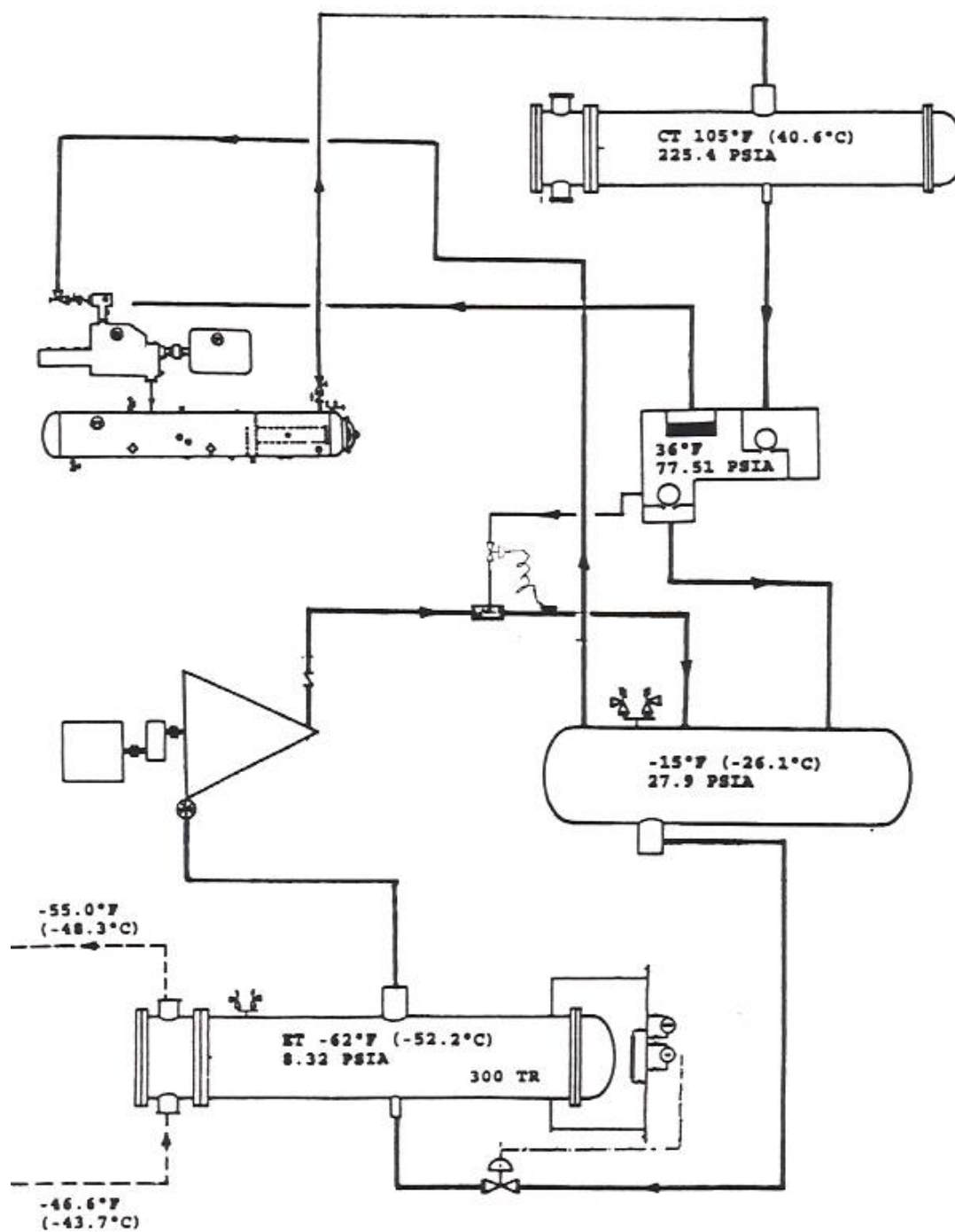


Figure 8-6 Refrigerant Flow Diagram R-22
Centrifugal Booster / Screw Compound System

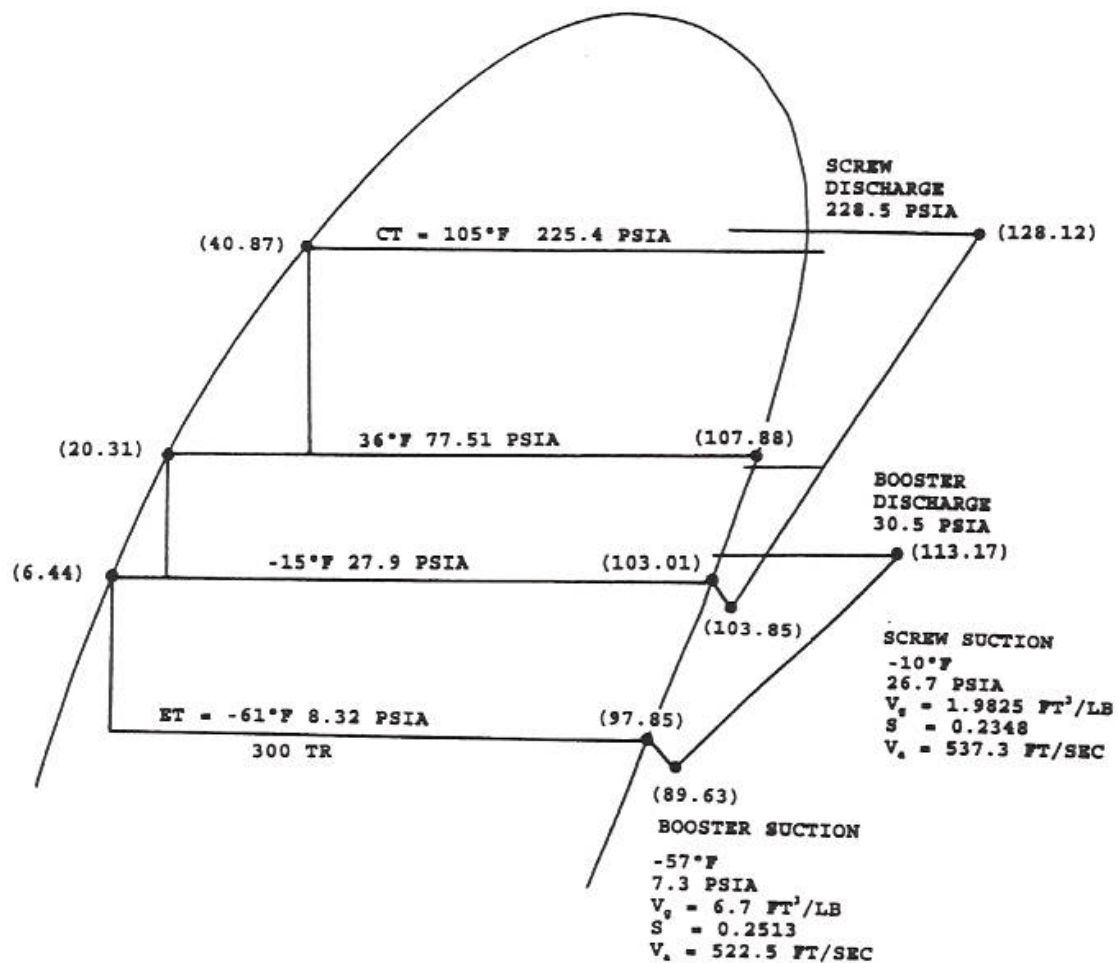


Figure 8-7 P-H Diagram Analysis R-22
Centrifugal Booster & Screw Compressor
Compound Refrigeration System

- When compound system cannot meet the design evaporative temperature as required.
- When the suction temperature is too low, the suction pressure cannot have a stable operation of the compressor.
- When the suction specific volume of the refrigerant gas is too large, the initial investment for the compressor and system are too expensive.

- D. Positive pressure requirement; the low side is to be operated above atmospheric pressure.
- E. When the overall temperature differential exceeds the practical range of one refrigerant.
- F. When the ET is below -80°F.

Cascade system is made up by two entirely separate independent refrigerant circuits as shown in Figure 8-8, each using a refrigerant appropriate for its temperature range. Heat is exchanged through a cascade condenser between the high side and the low side circuits. The high side is the high temperature system and the low side is the low temperature system.

The cascade condenser is a shell-and-tube heat exchanger to tie the low side and high side systems together. The refrigerant gas discharge from the low side compressor is condensed inside the tubes of the cascade condenser. The cascade condenser is also the evaporator for the high temperature system; the high side refrigerant liquid in the shell absorbs the heat from low side, evaporates and returns to the suction of high side compressor.

The evaporative temperature for the high stage for the cascade system depends on the evaporative temperature level of the low stage; it is usually in the range of 0°F to -30°F. The refrigerant use for the high stage circuit may be R-22, R-717, R-290 (Propane) or R-1270 (Propylene). The low temperature cycle usually uses a high pressure and low specific volume refrigerant, such as R-23, R-116, R-508B, R-170 (Ethane) or R-1150 (Ethylene); the sole purpose of using these refrigerants for the low side is to minimize the size and number of compressors.

The specific volume of the low side refrigerants is relatively small at the low temperature. However, the specific volume is getting larger and the pressure is getting higher when the temperature increases to the stand-by ambient temperature. In order to avoid high design working pressure (DWP) for the low side, an expansion tank is to be sized large enough to permit vaporization of all the refrigerant charge of the low side during the system is shut-down and warming up to the ambient temperature. The DWP of the system should be 300 Psig and the expansion tank volume should be sized large enough to limit the system pressure to be below 225 Psig.

Some case, a Hold-down System is provided for the cascade system. The hold-down system is a small refrigeration system to keep the low side cold when the cascade system is shut-sown. For safety consideration, it is suggested that the hold-down system should not replace the need of an expansion tank.

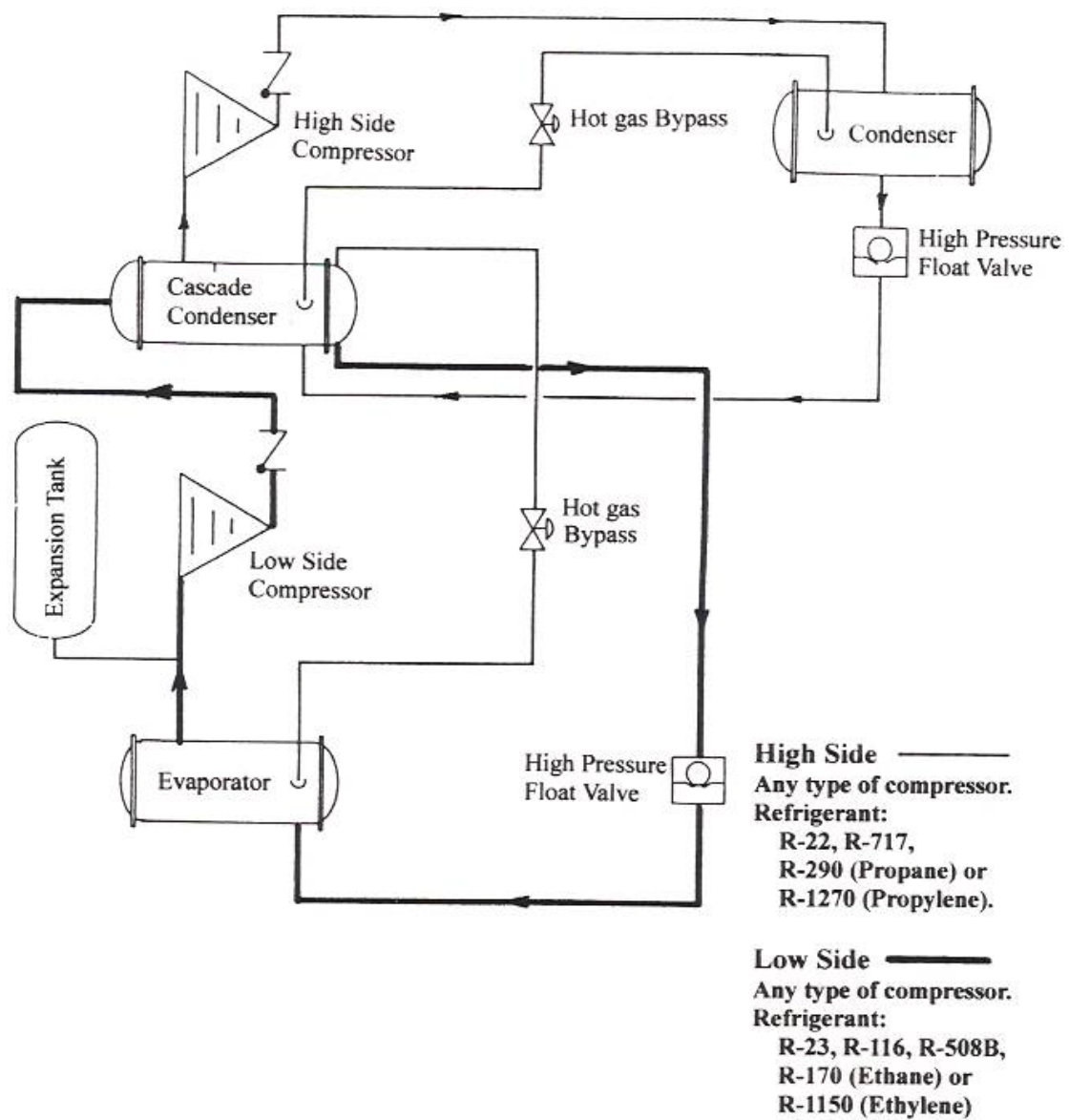


Figure 8-8 Typical Cascade Refrigeration System